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DEVELOPMENT OF CONCEPTUAL DESIGNS FOR AN IMPROVED M41 TYPE SPACE HEATER

BY
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This report describes an effort to develop conceptual designs for a new improved M41 type space heater. The design features of the current M41 were evaluated and ranked in order of importance. A literature search was performed to base possible design improvements using technology which was not available when the M41 was initally developed. Laboratory tests were performed on the M41 to develop a better understanding of the operation of the M41 heater. Finally, several design concepts were developed based on the results of the literature search and laboratory tests. It is concluded that a program to develop a new improved M41 heater, utilizing the results of the literature search and laboratory tests, has a high probability of success.

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PREFACE

This report identifies conceptual designs for an improved M41 space heater. It involves an analysis of the M41, literature search and laboratory tests yielding results which were used as the basis for developing the conceptual designs. The work described in this report was authorized by the US Army Natick Research and Development Center under contract No. DAAK-60-82-C-0054 and funded under project No. 1L162723AH98, Clothing & Equipment. This report uses US customary units because of the M41's application in US Tents. Project Officer was Joseph A. Mackoul.

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I. SUMMARY

1. Introduction and Background

This report summarizes work on the second task under Contract DAAK-60-82-C-0054 (Task 009) related to developing an improved M41 Tent Heater. In the first task, commercially available units which might be substituted for the M41 heater with a minor cost change were reviewed, and none was found suitable without significant design modifications. Therefore, in the task covered by this report, we evaluated ways of modifying the present M41 heater to improve its performance. The emphasis was on utilizing technology that was not available when the unit was initially developed. The work was broken down into five basic steps as follows:

- o partitioning the energy given off by the M41 heater into radiant and convective components from the flue and stove and estimating the performance of the unit in a tent;
- o evaluating the importance of the different design features of the M41 heater (i.e., ranking the design criteria for an improved M41);
- o searching the literature for information on which to base possible design improvements;
- o performing laboratory tests to develop a better understanding of the operation of the M41 heater; and
- o developing new design concepts.

At the conclusion of this work, recommendations for further work were developed. The balance of Section I presents a summary of results and our recommendations for further work in the development of a new heater. The remaining sections present the results in more detail.

2. Results

Energy Partitioning

The energy partitioning was carried out to develop a better understanding of the mechanisms that affect the performance of the stove from a comfort and efficiency viewpoint. These estimates (summarized in Table 1) show that the M41 heater delivers energy to the tent with an efficiency ranging from 71% at $-30^{o}F$ ambient to 62% at $32^{o}F$ ambient. A significant portion of the delivered energy comes from convection and radiation from the flue pipe. Convection from the flue pipe promotes temperature stratification in the tent which increases the heating load because temperatures in the upper portion of the tent are higher than necessary for comfort. Thus, it is estimated the actual

TABLE 1

Summary of Heater Efficiency Estimates

Ambient Temperature (oF)	Total Load, Btu/hr	Portion Associated with Stratification Btu/hr	Heater Input Btu/hr	Fraction to Tent %	Savings if No. Stratification	
					Btu/hr	%
32	23420	4700	37740	62	7580	20
-30	58880	4740	82900	71	6680	8

heating load is 8% to 20% greater than it would be if no stratification were present. This translates to a 8% to 20% fuel savings if stratification could be avoided.

Design Criteria

In order to learn what modifications would be considered improvements to the present M41 heater, a list of operational and design features of the present heater were reviewed. This review indicated that the primary problems to be addressed were smoking and plugging of the smoke stack with soot. It was felt that maintaining the quality operational simplicity, ruggedness, compactness and lightweight quality of the present unit was a paramount importance. The unit must be capable of burning all liquid and solid fuels burnt by the present unit. Any improved design must be nonelectric. Improving the safety of operation was also considered to be very important. Other factors of lesser importance were efficiency, type of venting, turn-down, thermal signature and noise. Some cost increase is permissible.

Literature Search

Available literature for the period 1920 through 1970, covering early design work on thermal vaporizing burners was reviewed in order to develop an understanding of the physical mechanisms operating in this type of oil burner. Four basic guidelines for reducing sooting tendency were identified, as follows:

- liquid fuel should be introduced in a way such as to promote rapid vaporization (e.g., thin film);
- maintain fuel-wetted surfaces above 750°F;
- delay combustion until adequate fuel-vapor air mixing occurs;
- promote carbon burn-out prior to contact with stove walls or flue pipe.

The bulk of the R&D activity for thermal vaporizing burners occurred in 1920 to 1940 prior to the introduction of the powered oil burner in the United States. A number of early designs addressing the low soot objective were identified. Most appeared to be relatively complex and probably would not meet the specific design goals of this project. A Dutch triple stage burner design comes closest to a commercially available unit. However, this unit is not appropriate for field use due to its complexity and sensitivity to set-up.

Laboratory Tests

A brief series of diagnostic tests was conducted on the M41 heater in order to observe performance factors that might shed light on the tendency of the M41 toward smoking and plugging of the flue pipe. These data were used to validate the theoretical results and to provide a baseline or starting point for developing design modifications. With both gasoline and diesel fuel, the

unit was found to have high smoke levels (between 5 and 9 on Bacharach Smoke Spot Device(R)*) at all fuel settings. Fuel turn-down rates of about 4 to 1 were found for both fuels although gasoline could be adjusted 5 to 1. At maximum fuel flow rates, excess air was reduced to zero on diesel fuel. A relationship between the flame and liquid pool size was developed which shows that the burning rate is directly proportional to flame thickness and fuel pool area and inversely proportional to the square of the height of the flame above the pool. These findings will be useful for further development work.

3. Design Concepts

Several design concepts were developed that embody the principles identified in the laboratory tests and the literature search. Figures 1, 2 and 3 give schematics of approaches which might be promising. Although additional laboratory work and design development is necessary before any of these schemes is actually implemented in a prototype unit, we feel there is a good probability of success for substantially reducing the sooting tendency.

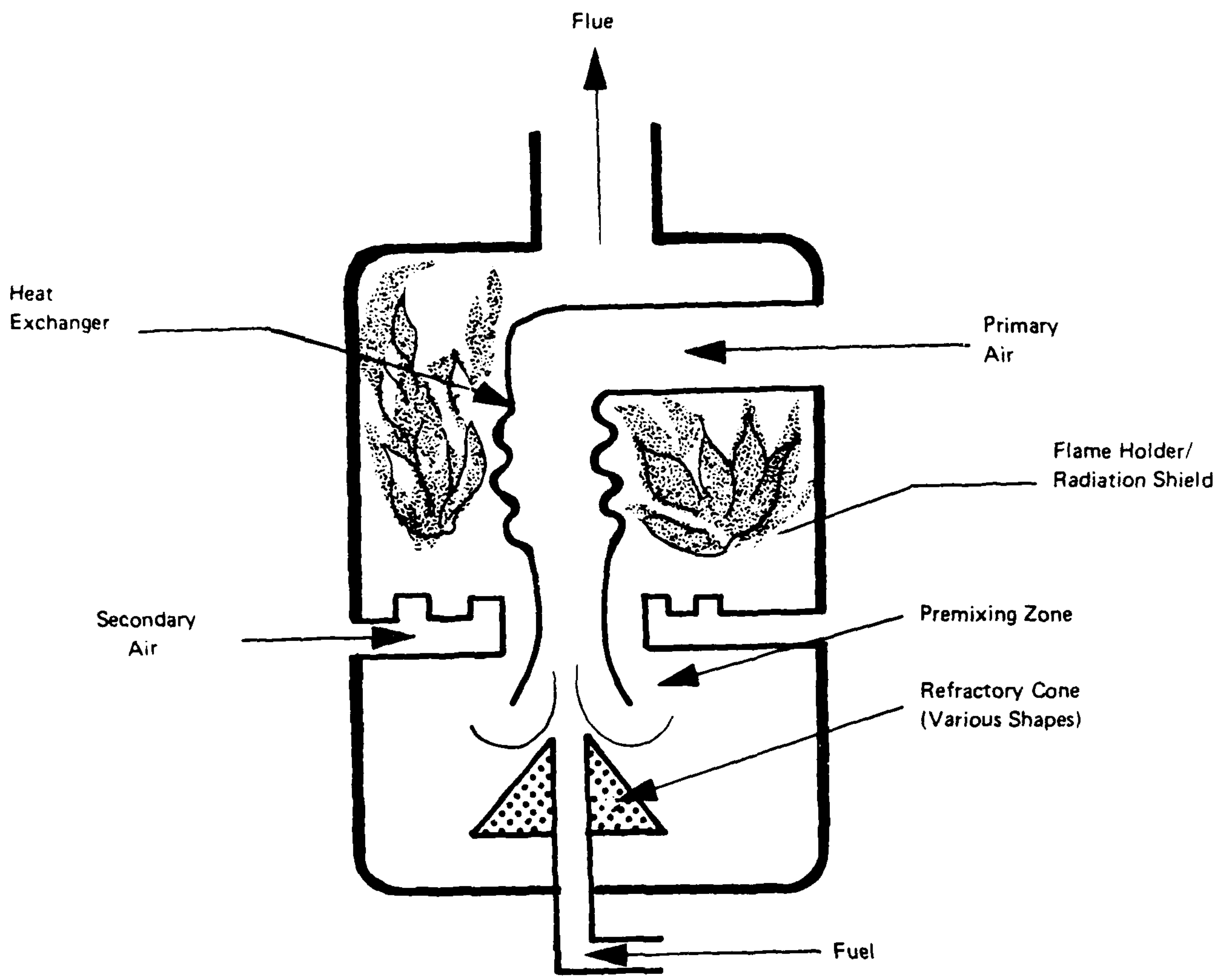
The major design features of the first concept (Figure 1) are (1) the stepped refractory cone designed to provide variable surface area with constant fuel thickness to accommodate different firing rates, and (2) a preheat zone to heat the incoming primary air. The refractory cone is designed to be more readily heated to an elevated temperature by radiation from the flame. Secondary air is introduced in a zone spaced away from the premixing zone to allow time for adequate mixing of air and fuel vapor.

The second concept (Figure 2), employs a mixing chamber designed to promote swirling and induced turbulence to mix the primary air and the vaporized fuel. Again, a premixed zone is provided with secondary air introduced in a zone removed from the primary mixing zone. A refractory liner is used in the primary chamber, to provide a hotter surface for rapid evaporation than otherwise possible.

The third concept (Figure 3), uses water to help atomize the fuel. This type of burner achieves performance similar to powered nozzle systems. However, a supply of water is required.

The precedent publications from which these design concepts were derived are noted on the figures. However, none of the units cited is the same as those shown in the figures. Rather, a combination of features have been employed to illustrate possible design approaches.

*Bacharach Smoke Spot Device is a registered tradename of Bacharach Instrument Co., Pittsburgh, PA. The use of trade names does not constitute an official endorsement or approval of the use of such items.



Precedents: Pizzi (USA)
Kermode's Mushroom Burner
Williams Heating (USA)
Oliver
Worthington
Oil Heat
Vulcan
Waltham

Figure 1. Design concept employing air preheat and refractory fuel cone.

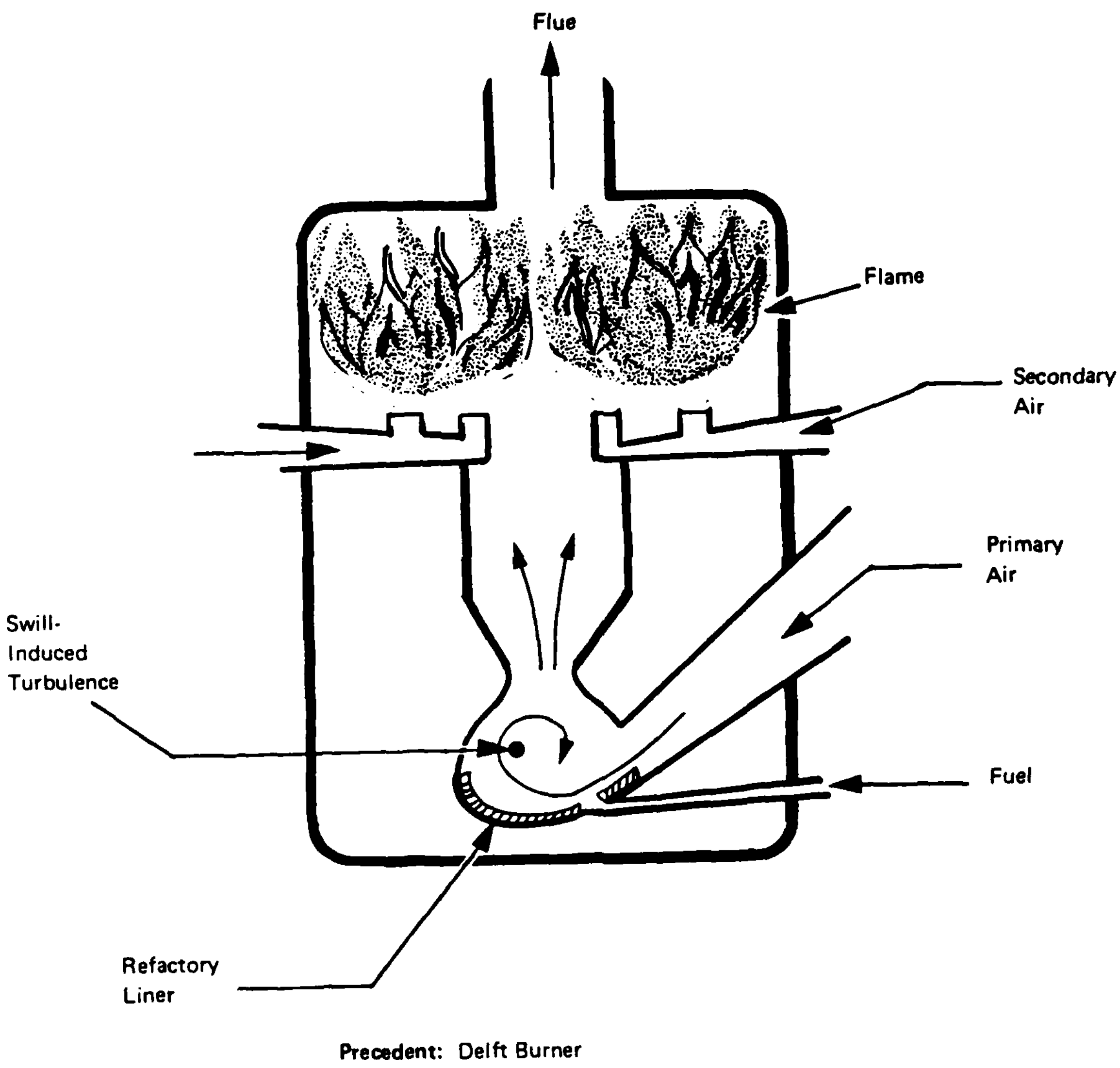
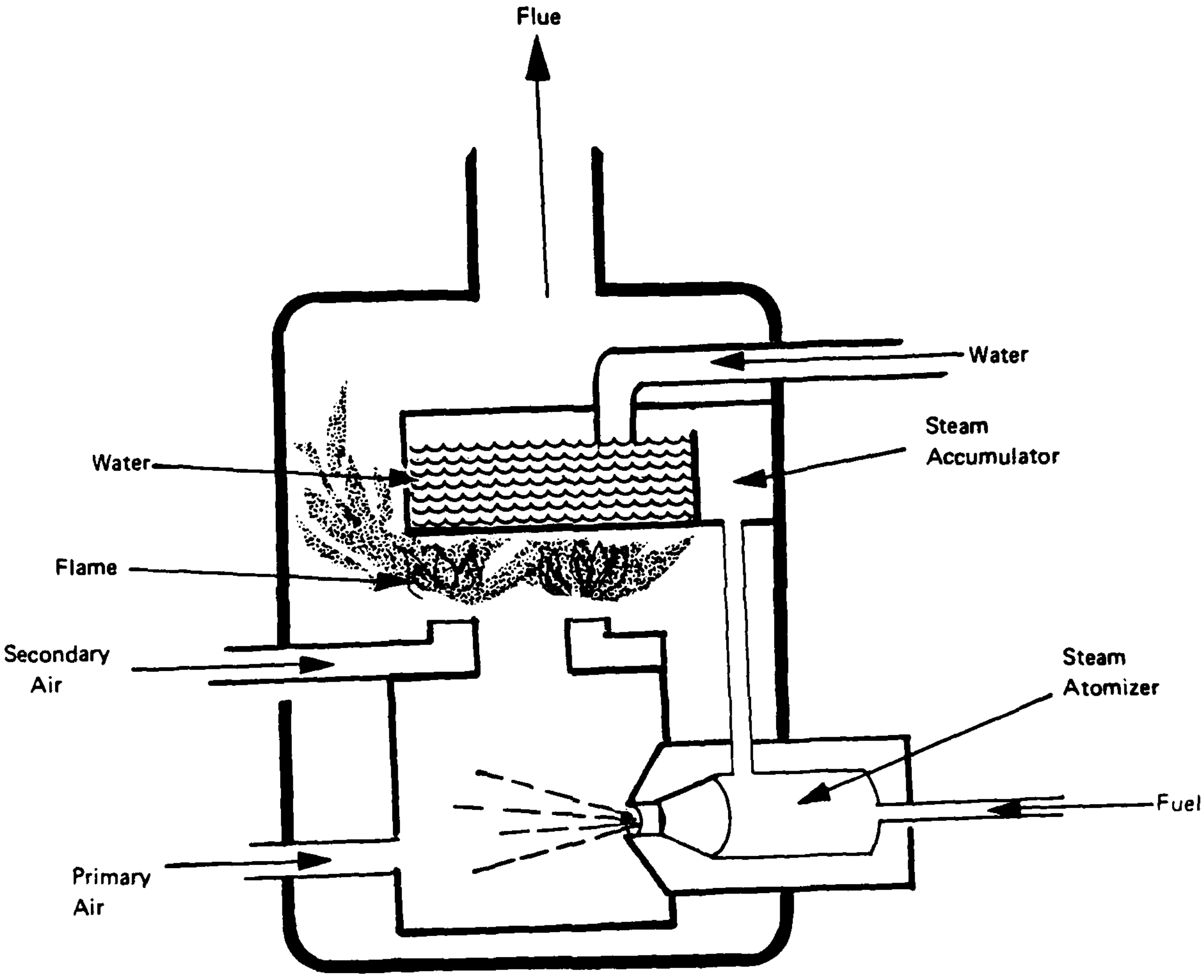


Figure 2. Design concept employing vortex premixing cell.



Precedents: Simplex (Austria)
Fitzpatrick (England)
Utinasa (Germany)
Filma (England)

Figure 3. Design concept employing steam atomization.

4. Recommendations

Based on this work, it is believed that substantial improvements can be made to the M41 unit while maintaining its basic simplicity, ruggedness, and multifuel capability. To continue the development of this product a three phase program is recommended as shown in Figure 4. In Phase I, Burner Research, a laboratory test rig would be designed and constructed to evaluate different burner concepts and to collect data that could be used in developing a prototype design. This test device would be designed to accommodate different pot burners, varying inlet air, and flame observations. Instrumentation would include thermocouples for measuring surface temperatures and monitoring the size of fuel pool, heat flux gauges, fuel flow measurements and combustion monitors. At the conclusion of this phase, sufficient design information would be on hand to develop prototype burner designs.

In Phase II, Burner Development, one or more prototype burners would be developed. The designs developed in Phase I would be fabricated and tested. Appropriate analysis and redesign, if necessary, would be carried out to develop a low-smoke burner prototype.

In Phase III, Stove Development, a complete demonstration stove would be developed, which incorporates the low-smoke burner from Phase II. This stove would include improvements in the fuel valve, enhanced heat transfer components, redesign of the flue pipe, and packaging of the unit.

It is believed that a manufacturer might possibly participate in Phases II and III on a cost-shared basis. The role of the manufacturer could be to supply prototype hardware to designs developed by the R&D contractor.

In summary, the results of this task indicate that the present M41 heater has definite shortcomings, which affect performance both from a safety and from an operational viewpoint. These can be corrected by employing design approaches identified in the literature and through laboratory tests.

It should be possible to develop a new unit which maintains the basic features of the present unit (ruggedness, low weight, nonelectric, multifuel, etc.) with a modest cost increase. We believe that this program has a high probability of successfully developing an updated version of the M41 tent heater, which uses modern technology to correct the problems of the older design.

II. LITERATURE SEARCH

1. Design Fundamentals for Smokeless Combustion of Fuel Oil

An investigation of available literature (1020-1970) covering early design work on thermal vaporizing burners has been conducted in order to appreciate better the underlying physical mechanisms of this type of oil burner, and to "stand on the shoulders" of earlier workers. Laboratory tests on the M41 stove confirm the manner in which the flame position and lazy oxygen mixing give rise to soot. Based on this literature review and the laboratory M41

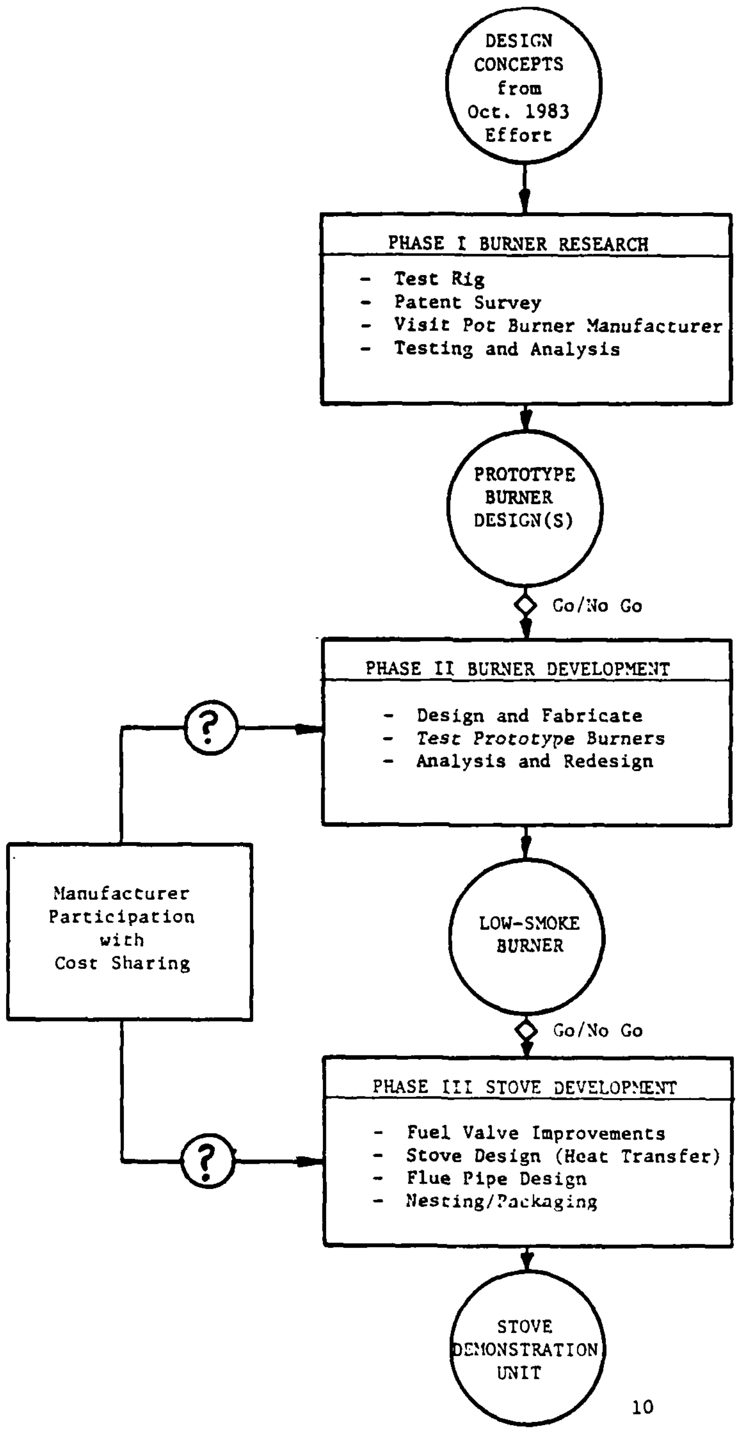


Figure 4. Recommended program.

tests, the following design fundamentals must be pursued in attempts to improve the M41 smoke characteristics:

a. The liquid fuel residence time in the vaporization zone should be minimized to prevent carbon deposits from the liquid. In practice, this means the liquid surface area should be maximized and the liquid film should be as thin as possible. Gun burners accomplish this by atomization but, with a vaporization burner, the liquid layer surface spreads by gravity and is limited by surface tension.

b. Carbon deposits from the liquid are minimized when the wetted surface is maintained above 750°F. The temperature of the wetted surface can be raised by the use of refractory, thermal conducting elements and adequate flame heat flux.

c. Carbon formation in the gas phase occurs when insufficient fuel vapor-air mixing occurs prior to combustion, leaving fuel-rich pockets of mixture. Upon combustion, fuel-rich zones are raised to high temperatures (1800-3000°F); and if held at high temperature for sufficient residence time, they produce carbon soot. Therefore, carbon formation in the gas phase can be minimized either by: (1) intensifying the vaporization/mixing process (through turbulence and atomization), or (2) prevaporization-premix or delayed combustion. Approach (1) is used in gun burners where electricity is used and firing rates are higher. The second approach (delayed combustion) is appropriate for vaporization burners. The basic idea of delayed combustion is to keep the vaporization zone sufficiently separate from the combustion zone to allow vaporization and premixing to occur. This is done largely by controlling the amount and flow pattern of primary air and controlling the heat flow received from the flame (baffles, flame position, etc.). Preheated air is also helpful because it promotes vaporization with greater flame standoff distance. The ideal is to produce a well mixed preheated gas consisting of fuel vapor, CO and inerts (CO_2, N_2) and to send this gas to the combustion zone. The flame front can be prevented from moving closer to the vaporization surface by restricting the primary air and placing a flame holder just upstream of secondary air introduction.

d. Some carbon forms in all oil flames, and, therefore, it is important to maximize carbon burn-out prior to contact with stove walls or the flue pipe. Carbon oxidation (in the combustion zone) can be maximized by secondary air "flame holder" designs, uniform mixing (swirl; turbulence), and adequate residence time at high temperature (heat transfer to the walls delayed until mixing and combustion is complete).

These are the principles or guidelines for designing low-soot thermal vaporization burners. In the next section, are examples of some early designs of vaporization burners to illustrate how these principles have been applied.

2. Early Design Attempts for Low-smoke Vaporizing Burners

Atomizing oil burners (available since about 1870) are technically far superior to the vaporizing type burner, basically because external energy is available to intensify the fuel mixing process. Atomizing burners are used in practically all marine and industrial oil-fired equipment. However, atomizing burners are relatively costly, noisy, limited to firing rates above 70,000 Btu /hr, and are most suitable for central-heating in residences with basements. Therefore, efforts to develop and refine the vaporization oil burner occurred in countries with small houses and noncentral space heating (England, france, Germany, Russia).

In the U.S., experimentation with the vaporization burner started with the 1893 World's Fair where the buildings were heated with oil as a novelty. By the 1929's, there were several manufacturers of vaporizing burners, mainly for residential heating and cooking in mild-climate areas where coal was more expensive (e.g., Pacific coast). However, this form of oil heating has always been limited nationwide because of the prevalence of basements (noise tolerance) and central heating. The obsolescence of vaporizing pot burners was assured when the API initiated a Research Program on Distillate Fuel Oil in 1960-1965, which perfected the modern retention-head atomizing oil burner. Ironically, this same Research Program included an extensive effort to develop a smokeless, automatic vaporizing burner, but the resulting prototype used twice the electricity of gun burners and was relatively complex.

One can conclude that the significant developments in <u>nonelectric</u> (gravity fed, natural draft, manual ignition) vaporizing oil burners occurred in the U.S. in 1910-1940, and in Europe in 1950-1960 (particularly in Holland and West Germany, where vaporization burners outnumbered atomization burners 6 to 1). It is this technology which forms the starting point for potential improvements to the M41. Some examples of this technology are given below.

The first vaporization burners to recognize the need to break up the oil and achieve greater surface area were developed in Russia. The "Astrakan" stove (Figure 5) was used in the 1860's for burning crude <u>oil</u> for home heating with a minimum of smoke. Refractory bricks served as a flameholder and radiated heat back to the fuel dish. Water was allowed to flow with the fuel oil, and the resulting differential in vapor pressure resulted in fuel oil atomization due to supercritical "explosions" of water vapor.

Another approach taken to enhance the vaporization was to feed the oil on a hot refractory surface, such as a step cone (Figure 6(a)) or a porous refractory "hot spot" (Figure 6(b)). In both of these burner designs (United States, c.1920), the importance of maintaining the wetted surface above 750°F was recognized, and preheated air was used.

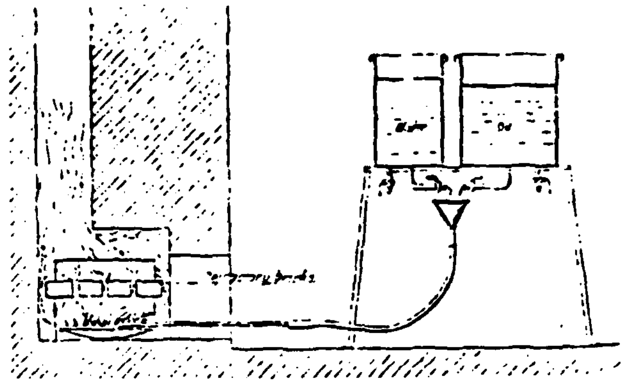
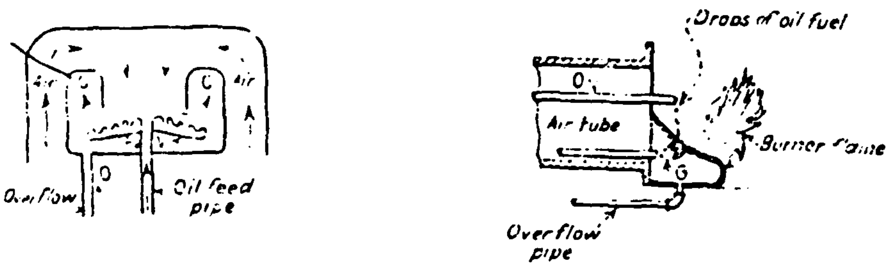


Figure 5. Astrakan Oil Stove.



(a) (b)

Figure 6. Early designs of vaporization oil burners.

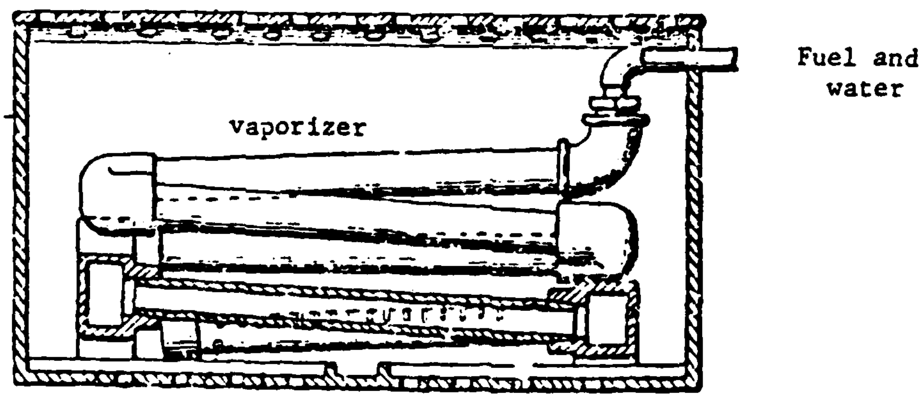


Figure 7. Hunt burner (Ref. 1).

Figures 7, 8, and 9 illustrate designs which take the prevaporization approach to the extreme of providing a separate chamber in which the fuel oil vaporizes due to heat flux from the flame. The key problem to avoid in this design is coking of the prevaporization chamber. The Hunt burner (Figure 7) claims to accomplish this by mixing fuel and water. The Hein-Weller prototype burner (Figure 9) employes continuous electric heating of the prevaporization chamber; this was the API sponsored design of the early 1960's, mentioned above.

Perhaps the most successful vaporizing burner design, to date, is the Dutch triple stage burner (Figure 10), versions of which have been tested by the Natick R&D Center. The success of this design, even for kerosene and diesel fuel, rests on separating the combustion process from the vaporizing fuel tray by a series of nested, perforated baffles.

These designs can be used as the starting point for developing prototype vaporizing burners that offer lower smoke and soot over a wider turndown ratio than the M41.

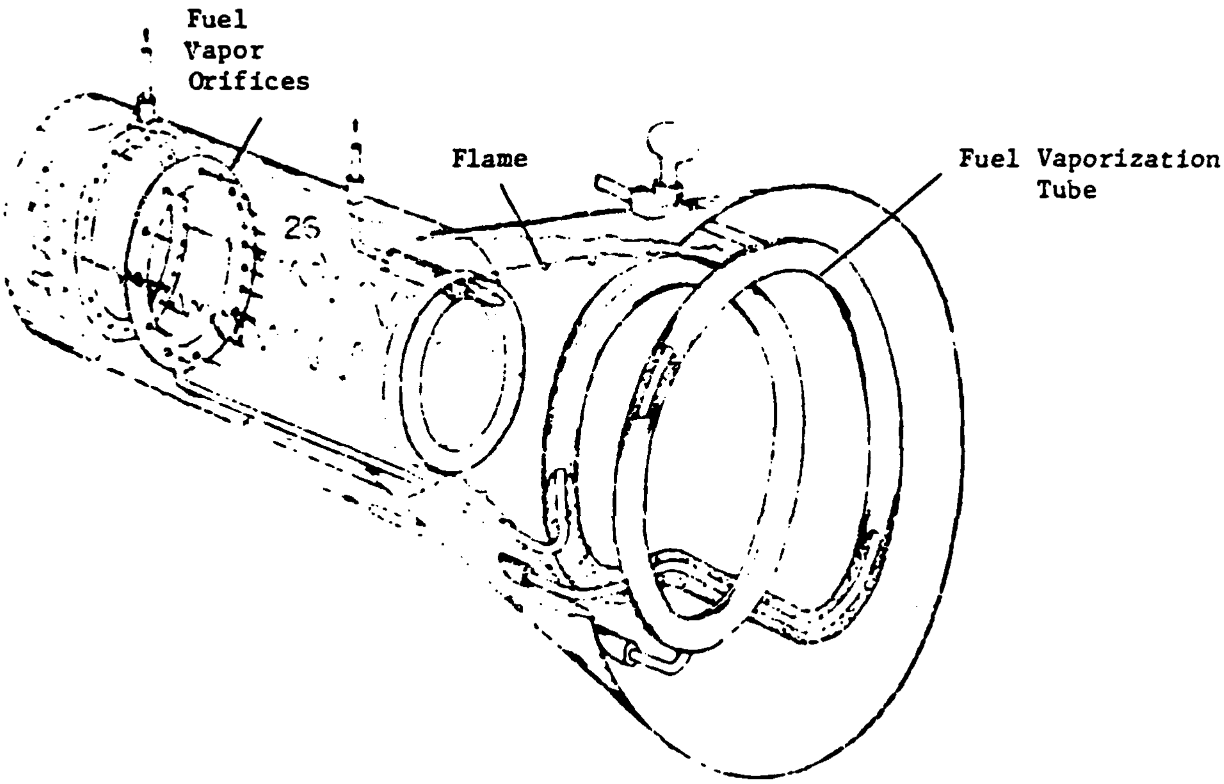


Figure 8. Roffe burner (Ref. 23).

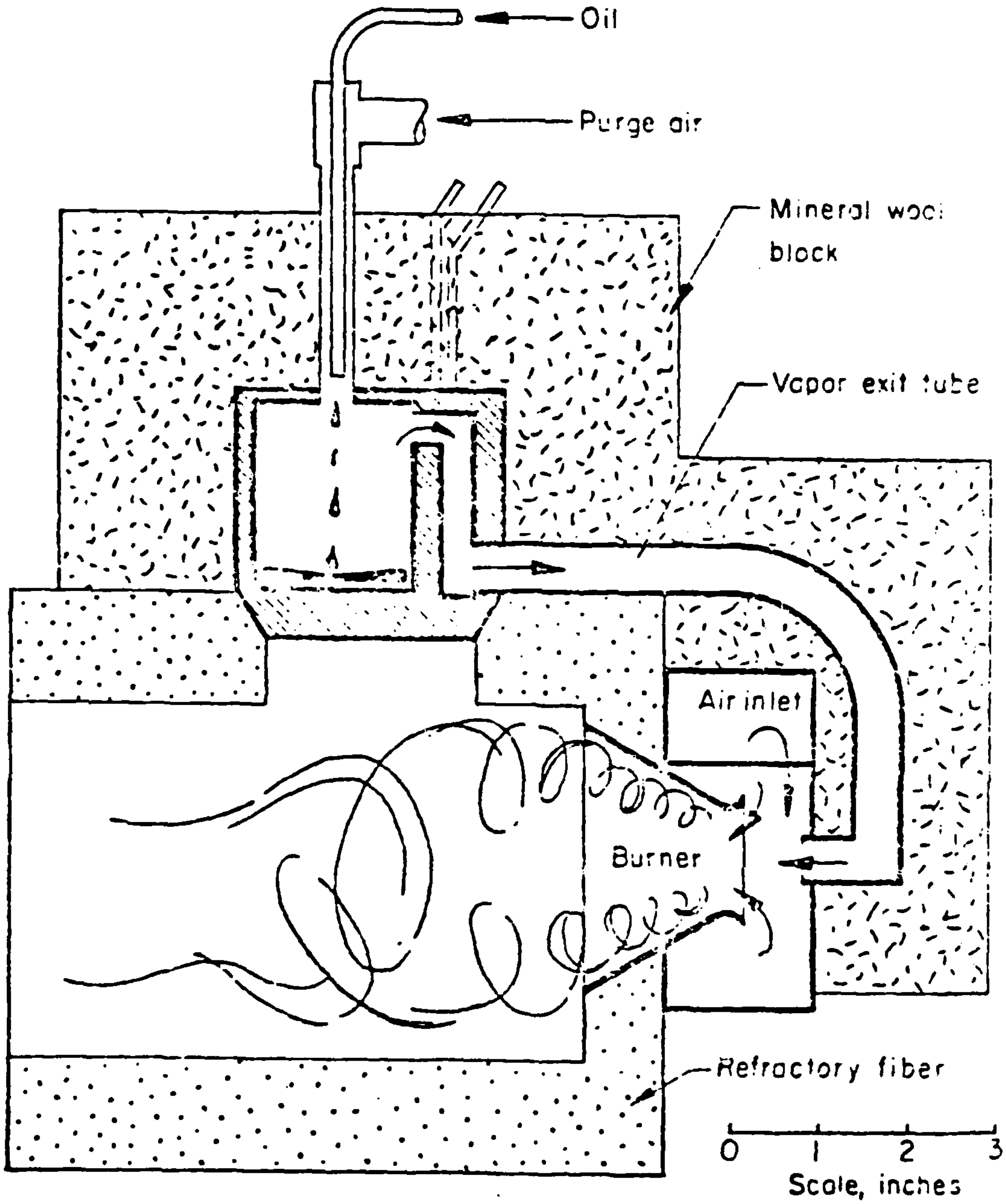


Figure 9. Hein-Weller prototype burner (Ref. 21).

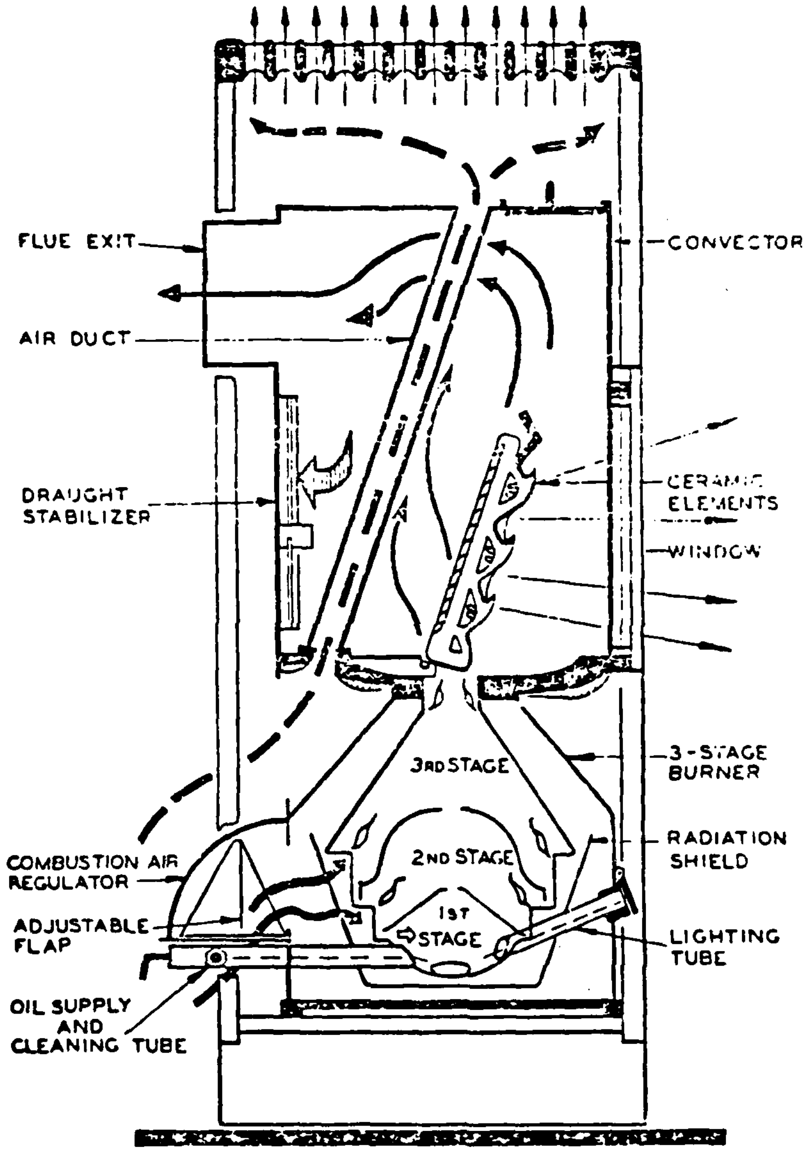


Figure 10. Arrangement of triple stage burner in heater employing convection and radiant heating (Ref. 19).

3. Technical Literature on Vaporization Oil Burners, 1920-1970

The following papers were identified by scanning Engineering Index and Chemical Abstracts for the period 1920-1970.

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24. Locklin, D.W., "Small Oil Burners - A Review of Some Recent Developments," ASME Paper 69-WA/Fu-5, 1969.

25. Roffe, G.A. and Trucco, H.A., U.S. Patent 4,008,041, Feb. 1977.

III. ENERGY ANALYSIS OF THE M41 STOVE OPERATING IN A TEMPER TENT

1. Introduction

The objectives of this section are: (1) to determine the typical heat loads of a Tent, Extendable Modular, Personnel (TEMPER) tent in cold climate conditions; (2) to show the extra heat load due to temperature stratification in the tent; (3) to estimate the required fuel firing rate for the M41 to meet this heat load (based on laboratory efficiency measurements), and (4) to develop a theoretical partitioning of the M41 heat flux by source (stove body and flue pipe) and by type (radiative and convective).

2. Methodology

A consolidated list of assumptions is given in Table 2. First, the heat losses from a 16' x 20' TEMPER tent were predicted for two outdoor temperatures, 32^{o}F and -30^{o}F, in order to obtain the required output of the M41 stove assembly. The fuel firing rate was then calculated by dividing the required heat output by the measured efficiency for the M41 stove assembly. The stove assembly consists of the stove body and the flue pipe, each of which gives off significant heat to the tent. The heat delivered by the stove body was calculated, based on the flue gas temperature at the entrance to the flue pipe and the excess air level. The heat delivered by the flue pipe was then calculated by substracting the heat delivered by the stove body.

TABLE 2.

List of Assumptions and Input Data for Energy Analysis of M41 Stove in TEMPER Tent

Assumptions

- Continuous operation (not cyclic)
- Linear vertical temperature profile inside tent

Input Data

- 8 ft height (stack and stove)
- Temperature-height profile of 52°F at floor; 90°F at ceiling
- Stove exit (stack entrance) temperature of 1740 and 1080°F for 83,000 and 38,000 Btu/hr input rate respectively
- Flue exit temperature of 970 and 750°F for 83,000 and 38,000 Btu/hr respectively
- Excess air level of 5% and 100% for 83,000 and 38,000 Btu/hr respectively
- Normal air infiltration rate of 1 air change/hr
- A_F = 320 ft^2 (16 x 20 ft tent)
- A_d = 246 ft^2
- A_w = 512 ft^2
- h_F = 0.21 Btu/hr ft^2 °F
- h_w = 0.59 Btu/hr ft^2 °F
- Wind Speed = 4.5 mph
- T_o = 32°F, -30°F
- Diesel fuel
- Emissivity of stove and flue pipe ε = 0.8
- Heat transfer coefficient of stove and flue pipe (natural convention) h = 1.8 Btu/h ft > °F

The balance of the heat input rate is lost in the flue gas waste heat. The extent of convective and radiant heat transfer was calculated assuming a natural convection heat transfer coefficient (h) of 1.8 Btu/hr ft^2 ^oF, and an emissivity ( ) of the stove and flue pipe wall of 0.8. The fraction of radiant heat delivered is given by equation (1).

$$\frac{Q_r}{Q_r + Q_c} = \frac{\sigma \varepsilon \left(T_w^4 - T_o^4\right)}{\sigma \varepsilon \left(T_w^4 - T_o^4\right) + h(T_w - T_o)} \quad (1)$$

where

Q = heat flux (Btu/hr ft^2)
T = temperature
σ = 0.17×10^{-8} Btu hr^{-1} ft^{-2} $^oR^{-4}$

subscripter
r = radiant
c = convective
w = wall
o = ambient

The tent heat loss characteristics were calculated from the heat loss data presented in reference (3). The temperature stratification inside the tent was assumed to be 4.2^{o}F/ft (52^{o}F at the floor to 90^{o}F at the ceiling) (see Figure 11). An indoor temperature of 65^{o}F at three feet elevation was chosen as the reference comfort temperature. The expression for heat loss through the tent wall was then calculated according to the following equations:

$$Q_W = \int_0^{A_W} h_W \quad (T - T_o) \; dA_W$$

$$Q_D = h_D A_D \quad (T - T_o)$$

$$Q_f = h_f A_f \quad (T - T_o)$$

where

h = heat transfer coefficient (Btu/hr ft^2 ^oF)
A = area of tent pertaining to subscript (ft^2)

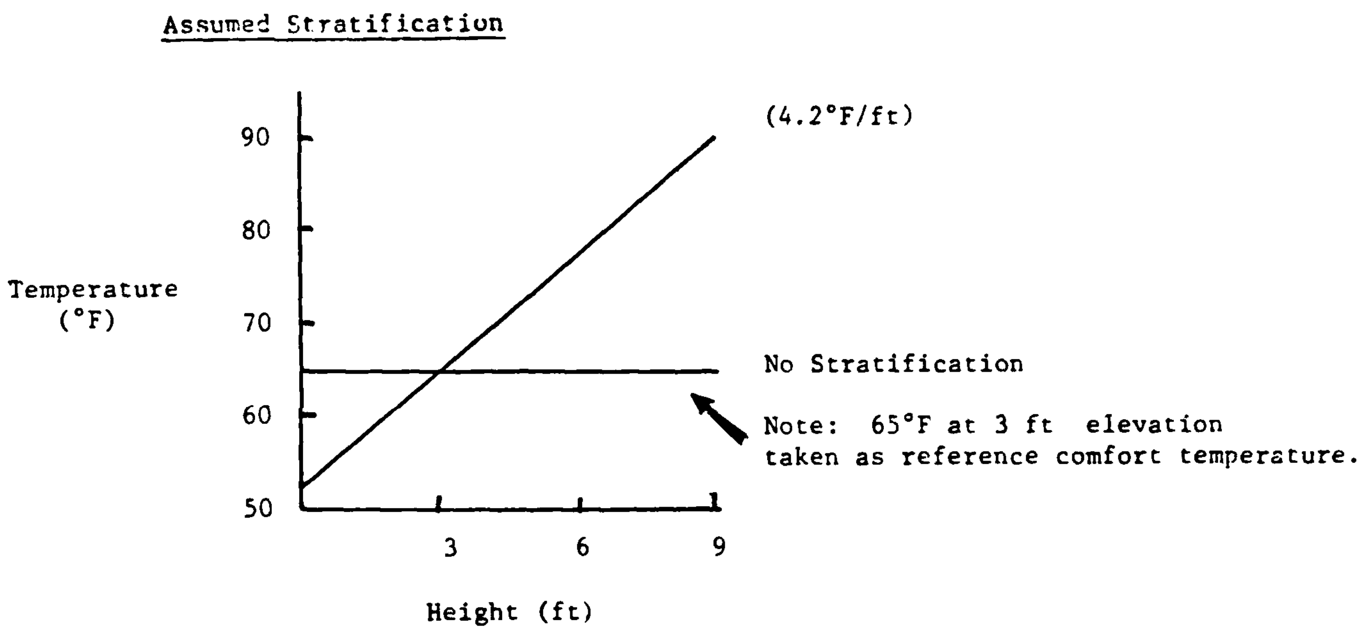


Figure 11. Method for estimating effect of thermal stratification on convective and conductive heat load.

T_o = outdoor temperature (^oF)
Q = heat loss through portion of tent pertaining to subscript (Btu/hr)

Subscripts
w = wall
D = roof
f = floor

A detailed calculation of the tent heat losses through the wall, ceiling, and floor, with and without thermal stratification for 32^{o}F and-30^{o}F outdoor temperature is presented in Table 3.

Stove induced infiltration was calculated by using the combustion air flow rates presented in Section V. The normal infiltration was calculated assuming one air change per hour inside the tent. The heat loads associated with infiltration rates were calculated according to equation (2).

$$Q_I = \dot{M}_{air} C_p (T - T_o) \quad (2)$$

where

$\dot{M}_{air}$ = mass flow of air (lb/hr)

C_p = specific heat capacity of air (Btu/lb^{o}F)

T = indoor tent temperature (^{o}F)

T_o = outdoor temperature ($_o$F)

Q_I = heat loss due to infiltration (Btu/hr)

The resultant energy partitioning for 32^{o}F and -30^{o}F outdoor temperature is shown in Figures 12 and 13 respectively. At 32^{o}F, outdoor temperature, the total heat load on the tent is 23,420 Btu/hr. To satisfy this load, the M41 must fire at 37,750 Btu/hr with a corresponding efficiency of 62%. About 50% of the heat input rate is delivered to the tent directly from the stove walls. Approximately 12% is delivered by the walls of the flue pipe. The remaining 38% of the fuel input energy is lost as flue gas waste heat. Radiant heat transmission accounts for 60% of the total heat transfer from the stove and 50% for the flue pipe.

The normal tent heat load (excluding infiltration) accounts for 45% of the fuel input energy. An additional 8% of the fuel input energy heats normal infiltration air bringing the total normal tent heat load to 18,300 Btu/hr. Heat loads from thermal stratification and combustion air account for 13% and 1% respectively, of the fuel input energy.

At -30^{o}F, the stove operates at full capacity to maintain the 65^{o}F comfort level at a three-foot evaluation inside the tent. The heat delivered by the stove body is 50% of the input energy, with radiant heat transmission accounting for 72% of this heat component. The heat delivered by the flue

TABLE 3

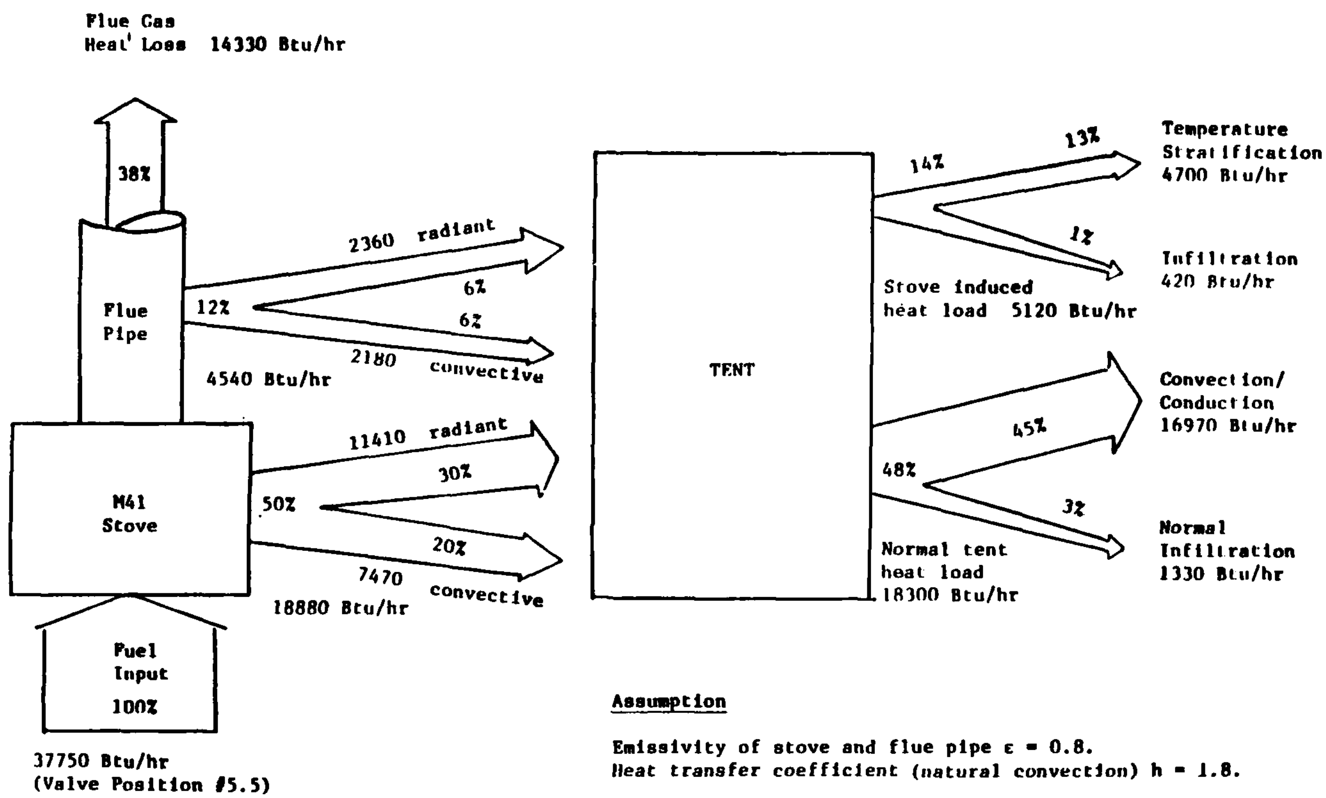
Detailed Calculation of Tent Heat Losses with and without Thermal Stratification

Component of Heat Load	T_o - 32 °F		T_o = -30 °F	
	$\frac{dT}{dx} = 0$	$\frac{dT}{dx} = 4.75$ °F/ft	$\frac{dT}{dx} = 0$	$\frac{dT}{dx} = 4.75$ °F/ft
$\dot{Q}_W = 0.59\ (171)\ (T - T_o)$	3,320 Btu/hr	2,690 Btu/hr	9,590 Btu/hr	8,960 Btu/hr
	3,320	3,960	9,590	10,240
	3,320	5,210	9,590	11,520
	9,960	11,870	28,770	30,720
$\dot{Q}_D = 0.59\ (246)\ (T - T_o)$	4,790	8,420	13,790	17,420
$\dot{Q}_F = 0.21\ (320)\ (T - T_o)$	2,220	1,380	6,390	5,550
Subtotal	16,970	21,670	48,950	53,690
Normal Infiltration	1,330	1,330	3,790	3,790
Stove Induced Infiltration	420	420	1,400	1,400
Subtotal	1,750	1,750	5,190	5,190
Total	18,720	23,420	54,140	58,880

3. <u>Conclusions</u>

1. At 32°F outdoor temperature, the temperture stratification inside the tent caused 25% addition load.

2. At -30°F, temperture stratification causes 10% additional load.



Assumption

Emissivity of stove and flue pipe $\varepsilon = 0.8$.
Heat transfer coefficient (natural convection) $h = 1.8$.

Figure 12. Energy flow partitioning for M41 stove operating in tent.
(32°F Ambient Temperature)

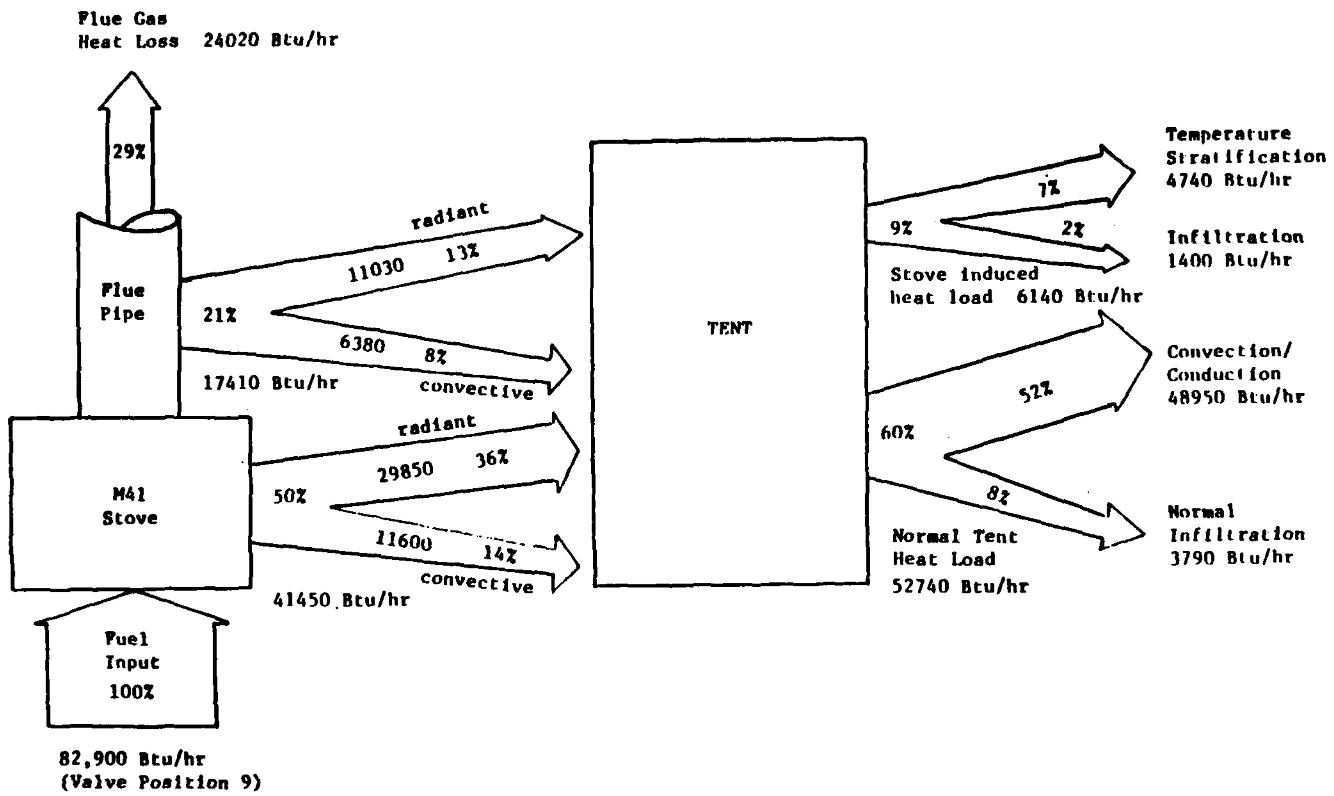


Figure 13. Energy flow partitioning for M41 stove operating in tent.
(-30°F Ambient Temperature)

pipe is 21% of the input energy, with radiant heat transfer accounting for one half of this heat component. The normal tent heat load (excluding infiltration) absorbs 52% of the energy input. Normal infiltration accounts for another 8% of the input energy. Stove-induced heat loads by thermal stratification and infiltration air account for 7% and 2%, respectively, of total input energy. Neglecting the heat transferred from the stove to the tent to satisfy stove-induced loads, the useful efficiency of the stove is about 60%.

IV. RANKING OF DESIGN CRITERIA FOR DEVELOPING NEW CONCEPTS

In order to learn what would be considered an improvement to the present M41 tent heater, various operational and design features of the present heater were reviewed with the Natick project team. Based on this review, different areas are identified for improvement and their relative importance. In addition, the specifications for the present unit were reviewed to establish which were fixed and which had some room for change. The matrix used to guide these discussions is shown in Table 4. Also shown in Table 4 is the ranking of the relative importance of various heater characteristics, and comments as appropriate. These design considerations will be useful in two ways:

(1) to guide further investigation and suggest the most fruitful directions;

(2) to use as criteria or standards against which to evaluate prototype design concepts.

Specific heater characteristics are discussed below.

1. Input Rate

The present unit is rated at 80,000 Btu/hr input at the maximum setting. This is equivalent to about 40,000-45,000 Btu/hr output rate from the stove body, excluding the flue pipe. Lab measurements indicate that the hot flue pipe gives off an additional 17,000 Btu/hr at maximum input rate. The total output rate of the M41 is therefore 55,000-60,000 Btu/hr at maximum firing rate. Although this maximum setting is appropriate for the general purpose tent in basic cold weather (-5°F to 20°F), it was felt that, in most cases the output was higher than necessary. In particular, for the new TEMPER tent with cotton liner insulation and reduced infiltration, lower outputs rates would be acceptable for cold weather, perhaps as low as 35,000-40,000 Btu/hr. However, for design purposes, a maximum output rate equal to the M41 (on the order of 55,000 to 60,000 Btu/hr) will be used as a guideline. It should be noted that this output rate is consistent with input rates as low as 70,000 to 75,000 Btu /hr if efficiency improvements can be made.

2. Turndown

The turndown rate, which is 5 to 1 for gasoline and 4 to 1 for diesel fuel oil, is acceptable. The lowest setting gives an output heating rate of about 12,000 Btu/hr, below which tent heating is not considered really necessary. Lab measurements show that on setting #9, the fuel flow rate was 38 cc/min and on setting #1 the fuel flow rate was 9 cc/min, giving a 4.2 to 1 turndown ratio for diesel fuel oil. With gasoline, although the maximum firing rate intended to be restricted to setting #7 to prevent overheating the flue pipe, in practice, the present design does not prevent people from misusing it (exceeding setting #7); as a result, gasoline fuel flow rates as high as 50 cc /min are observed. Ideally, the system should be designed so that overfiring with gasoline is not possible, and improvement in this area is desirable.

TABLE 4

Design Considerations/Criteria for Developing New Concepts

Consideration/Criteria	Present Unit M41	Comments, Acceptable Values, Etc.	Relative Importance 0-10 Where 10 is Very Important*
Input Rate	80,000 Btu/Hr	Realize that TEMPER tent has lower requirement but heater will be used for all tent types.	7
Turn Down	5/1 (except gasoline)	Below about 12,000 Btu/hr no heat is needed 16,000 - 80,000 Btu/hr (5-6 to 1)	5
Multifuel	all liquid and solid	Burning liquid fuel with trend to diesel fuel. Solid fuel for emergency backup.	9
Electric/Nonelectric	nonelectric	Sealed environment good but must be nonelectric	10
Vented (outside air forced draft)	yes	Just must be safe and nonelectric. Good to use outdoor air.	3-5
Simplicity	high		10
Ruggedness	high		10
Cost	$150	$250 not out of question	5
Smoke Level/Carbon Deposition	high	Smoke signature not a major problem. Soot deposition and plugging is the major problem.	10
Thermal Signature	500-900°F	Soot accumulates	2
Noise	low	M41 is quiet	5

TABLE 4 (cont'd)

Design Considerations/Criteria for Developing New Concepts

Consideration/Criteria	Present Unit M41	Comments, Acceptable Values, Etc.	Relative Importance 0-10 Where 10 is Very Important*
Safety	medium-low	Gasoline restart, hot surface constant level valve, leaking.	10
Efficiency	60-70%	Acceptable. Some loss is tolerable.	3
Physical Size	18.5" dia x 17" high	Desire small as possible.	9
Weight	35 lbs	M41 is "portable	9
Self-Storage Components	no		3-5
Other: Valve design	Fuel metering valve	Viscosity difference between fuels makes heating level hard to control	7

*Based on Natick Project Team comments.

3. Multifuel

The new unit should be capable of burning typical liquid fuels (diesel, kerosene, gasoline, and jet fuel) and solid fuels (coal and wood). Partial disassembly or removal of diffusor components is permissible in order to burn solid fuels.

4. Electric vs. Nonelectric

The unit must be nonelectric.

5. Type of Venting (Vented, Direct Vent, Forced Draft)

The venting system, per se, is not important as long as there is no added hazard to the tent occupants from exhaust gases. In this application, the modest fuel savings expected from direct vent is unimportant (see below, under Efficiency). However, a direct vented unit would be desirable for tents equipped with the passive option for chemical warfare defense, because it could work in a sealed environment independent of the air supply to the tent. This would prolong the usefulness of filters by reducing the amount of air drawn through the filters.

6. Simplicity and Ruggedness

As in the present M41 unit, both of these characteristics are essential.

7. Cost

The cost of the new unit is not a driving factor. Increasing the unit price from $100 or $150 to $250 would not be out of the question, provided significant improvements were achieved.

8. Smoke Level

The visible signature due to high smoke level is not considered a problem. However, carbon deposits and plugging of the stack pipes, soiling of the tents with soot and unburned hydrocarbons, and emission of "sparks" (glowing carbon particles) are major problems which must be addressed in the redesign of the system. Frequent manual cleaning of the stove is also objectionable. The problem is particularly severe with diesel fuel. However, with the trend to using diesel fuel for all field operations, the heaters will be increasingly fired with diesel fuel. Solutions to the high-smoke problem are seen as the highest priority objective of an improved heater design.

10. Thermal Signature

Reducing the thermal signature is not important, since these heaters are generally used in an area where there are other devices with thermal signatures equal to or greater than that of the stove.

11. Noise

The M41 noise level is acceptable. Any noise level increase of a new design should be minimum.

12. Safety

Improving the safety is an important design goal. The major problems with the use of diesel oil are ignition and burning of carbon deposits and associated fires started by sparklers and explosion hazard if a warm heater is relighted. The major problems with the use of gasoline include: (a) leaking fuel; (b) high temperatures in the stack, which may ignite the tent fabric or melt the stove components; (c) inability to control the flow adequately with the valve; (d) lack of a fail-safe design and (e) hazards due to difficulties in restarting the unit after it is stopped.

13. Efficiency

Improving the efficiency of the unit has a low priority because the heaters use relatively small amounts of fuel. In fact, if the heaters were to be modified to address other important factors, but the efficiency were reduced somewhat, this would be acceptable.

14. Physical Size

It is important that the physical size (both dimension and weight) be maintained as small as possible. Little change from the present unit size would be desirable, unless it can be made more compact.

15. Self-Storing Components

If possible, without sacrificing the other features identified as important, it would be desirable to have a design that includes provision for "nesting-type" storing all the stove components to protect them from damage during transit and to provide more compact storage.

16. Heating Ability and Tent Thermal Stratification

Although fuel use is not important, the M41 heater has been criticized for not providing uniform space heating, because it provides only adequate heat for occupants in close proximity to fit. Some improvement is desirable in the ability of the new heater to project heat uniformly and minimize stratification effects.

17. Other

A new valve design which provided better control, independent of the viscosities of the different fuels used, would be a desirable feature.

In summary, the major emphasis of the new design should focus on the ability to burn diesel fuel safely without generating high levels of soot and smoke

which tend to plug up the stack and deposit combustible residue on the tent. It is important that this be done with a design which is nonelectric, simple, rugged, safe, and has a small physical size and weight. Other features are less important.

V. LABORATORY TESTS OF THE M41 THERMAL PERFORMANCE CHARACTERISTICS

1. Introduction

A brief series of diagnostic tests were run at Arthur D. Little on 9/30/83 and 10/14/83 using gasoline and diesel fuel in order to measure thermal performance characteristics such as:

- o Exesss air at various valve settings;
- o Total heat absorbed (radiated and convected) by the stove and flue pipe;
- o Fuel flow at various valve settings, including turndown ratio;
- o Overall efficiency for the stove system (including flue pipe); and
- o Smoke levels.

2. Summary of Key Results

Excess Air: Increases from zero at maximum fuel flow to 125% at valve setting #4 (35% of maximum fuel flow) for operation on diesel fuel. Total air flow is relatively constant, because of buoyancy-driven natural draft.

Total Efficiency: Ranges from about 70% at maximum firing rate to about 60% at minimum firing rate. The flue pipe makes an essential contribution to these efficiency figures as it is responsible for 10% to 30% of the heat extracted from the hot combustion gases. In other words, the stove alone has an efficiency in the 50% to 55% range, as reported earlier by Natick R&D Center.

Fuel Flow Rate and turndown: At maximum setting (#9), the fuel flow rate was equivalent to 83,000 Btu/hr with diesel fuel; with gasoline the input rate was 25%-30% higher (100-110,000 Btu/hr). At the maximum gasoline setting with the laboratory lights off, luminous gases could be seen flickering at the flue pipe exit and the entire stove and flue pipe glowed red, suggesting that continuous operation might be hazardous.

The useful turndown ratio was about 4/1 for both fuels, with the minimum setting (#1) equal to about 20,000 Btu/hr for both fuels. Gasoline could be run up to 5/1 but this is inadvisable. The measured output rate is about 60,000 Btu/hr maximum and 12,000 Btu/hr minimum.

Smoke Levels: Ranged from 5-9 Bacharach Number at all settings for all fuels, showing the high smoke characteristic of this design.

In addition to these efficiency and smoke tests, a few measurements of the liquid fuel pool area and approximate flame height were made. This was done in order to understand the relationship between radiative heat flux, evaporation rate, and burning rate. It became clear that the area of the liquid fuel pool increases substantially when the valve setting is increased. Mixing of fuel and air takes more time (and more stove volume) so that flame sits higher in the stove. Because radiation falls off as the inverse square of the distance, the radiative flux per unit area, received by the pool, is actually lower for higher firing rates. This information will be very useful in seeking improvements to the M41 stove.

3. Description of Laboratory Tests

Stove Setup

The M41 stove was set up in a well-ventilated laboratory (70°F ambient) so that the total height of the stove and flue pipe was eight feet. A slight draft at the stove pipe exit (.02" water column) was induced by an exhaust hood to vent the flue gases to the outside. This draft is thought to simulate the draft conditions of a TEMPER tent in cold outside air with moderate wind. The stove rested on a concrete floor for all tests.

Measurements

Figure 14 illustrates the stove setup and location of temperature measurements and flue gas sampling on the M41 stove. The stove and flue pipe were instrumented to provide the following measurements.

Fuel Flow Rate.

The fuel was placed in a 1000 cc graduated cylinder which was located 30" above the stove. The graduated cylinder had a 1/8 NPT hole in the bottom through which the fuel flowed by gravity to the float valve. The time durations for the stove to burn 300-500 cc of fuel were recorded with a stopwatch.

Stove Body Temperature

The stove body temperature was measured with 22 gauge Type K chromel-alumel thermocouples on three locations shown in Figure 14. A sheet metal screw tapped into the side of the stove held the thermocouples in place.

Flue Gas Temperature

The same type thermocouples were mounted in the center of the flue pipe at the bottom, middle, and top of the pipe (see Figure 14). The flue temperature measurements were corrected for radiation errors as shown in Table 5.

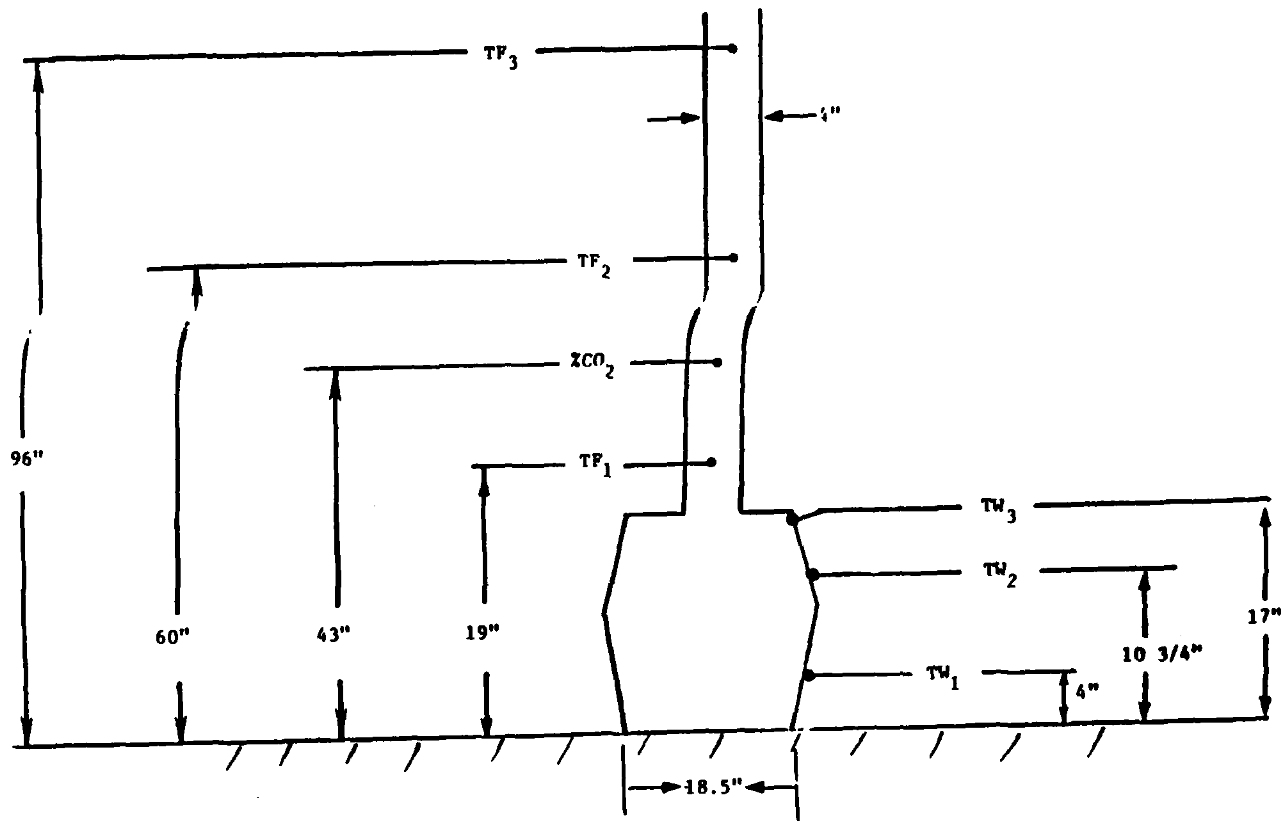


Figure 14. Location of temperature measurements and flue gas sampling on M41 stove.

CO_2 Concentrations and Smoke Number

Flue gas was sampled for smoke and CO_2 concentrations at an access hole 26" above the stove. Smoke measurements were taken with a Bacharach Smoke Spot Device and CO_2 concentrations were measured with a Horiba Infrared$^{(R)*}$ gas analyzer.

All temperature measurements were monitored by a data logger and displayed every 10 seconds.

4. Discussion of Test Results

Diesel Fuel

The test results on the M41 are summarized in Tables 6 and 7. On diesel fuel, the M41 exhibited at 4 to 1 turndown with a maximum input rate of 83 MBtu/hr at valve position 9. At this maximum firing rate, the stove body temperature ranged from 400°F at the bottom of the stove, to 950°F at the top of the stove. The flue gas entered the pipe at 1710°F-1740°F, dropped in temperature to 990°F-1050°F at the mid-section of the pipe, and exited the flue pipe at 900°F-920°F.

The temperature measurements at the base of the flue of the stove agree quite well with the data listed in Reference 2 (Section II). However, the measured excess air levels of 0-5% for high firing rate were considerably lower than the 40-50% excess air levels reported in Reference 2 (Section II). Attempts to increase the excess air level by increasing the draft to 0.05" water column had little effect on the overall excess air level. As expected, since the excess air levels were lower, the smoke emissions were higher. The smoke number for all tests ran on #2 diesel fuel were consistently above #7 with spot # of 9 or greater for the high firing rates. Based on the test results, it appears that the *flow rate of combustion* air was inadequate to ensure complete combustion at valve position #9. The mass flow rate of air through the stove is relatively constant, ranging from 50-65 lb/hr (see Table 7). The efficiency figures presented in Table 6 were calculated by the stack loss method using:

$$n = 1 - \frac{C_p(T_{F_3} - T_o)\ 1 + \frac{A}{F}}{¢H_c}$$

where

n = efficiency (%) by stack loss method
$¢H_c$ = higher heat of combustion of fuel (Btu/lb)

TABLE 5

Thermocouple Radiation Correction for Flue Temperature Measurements in M41

The bare thermocouple positioned in the flue pipe is heated by convection,

$$q_{conv} = hA(T_g - T_b)$$

where

h = convective heat transfer coefficient (Btu/hr ft^2 ^oF),

A = area of bead (ft^2),

T_g = actual gas temperature ^oF,

T_b = bead temperature ^oF,

and cooled by radiation:

$$q_{rad} = \sigma\varepsilon A\ (T_b^4 - T_w^4)$$

where

σ = Stefan Boltzman Constant, 0.17 x 10^{-8} Btu/hr ft^2 $^oR^4$

ε = emissivity of bead, 0.5 assumed

The resulting difference between the true gas temperature and bead temperature can be expressed by equating the above two equations and solving:

$$\text{Thermocouple Correction} = T_g - T_b = \frac{\sigma\varepsilon}{h}(T_b^4 - T_w^4)$$

Thermocouple Correction

Indicated $T_b(^oF)$	$T_w(^oF)$	$h\ (\frac{Btu}{hr\ ft^2\ ^oF})$*	$T_{correct}$	Actual T_g
1500	750	57	190^{o}F	1690^{o}F
800	414	37	45^{o}F	845^{o}F

*Derived from correlation for forced convection around spheres in Bird, Stewart and Lightfoot, John Wiley and Sons, New York, 1960, Transport Phenomena.

Table 6

M41 TENT HEATER TEST RESULTS

Fuel	Valve Position μ	Input Rate MBTU/hr (±0.2)	Fuel Flow CC/Min (±0.1)	TW_1 °F (±10)	TW_2 °F (±20)	TW_3 °F (±40)	T_{FLUE_1} °F (±50)	T_{FLUE_2} °F (±40)	T_{FLUE_3} °F (±40)	CO_2 % (±0.5)	EA %	Smoke Bacharach μ	Effic %	Output Rate MBTU/hr
Diesel Fuel Oil 10/14/83	1	19.7	9.0	-	-	-	-	-	-	-	-	-	-	-
	5	30.4	13.9	250	410	600	965	810	700	6.5	125 ± 12	7	63	19
	7	63.1	28.8	350	760	970	1740	1100	950	9.5	60 ± 5	8-9	60	38
	8	74.0	33.8	370	800	970	1740	1050	920	14.8	5 ± 5	9	72	53
	9	82.9	37.8	400	815	950	1710	990	900	14.5	0	>9	72	60
Gasoline 9/30/83	1	20.9	10.5	250	360	500	770	630	570	4.5	230 ± 20	8	61	13
	3	30.7	15.4	270	450	615	1060	800	710	6.2	150 ± 15	5	61	19
	4	45.2	22.7	250	520	720	1320	960	840	6.2	140 ± 15	6	55	25
	5	61.4	30.8	330	685	890	1575	1110	970	12.0	28 ± 5	7	68	42
	7	93.7	47.0	415	860	890	1770	1140	760*	--	Fuel Rich	>9	Incomplete Combustion	--
	9	100.4	50.4	360	855	1010	1585	980	810*	--	Fuel Rich	>9	Incomplete Combustion	--
	9	109.8	55.1	415	880	1010	1585	1010	750*	--	Fuel Rich	>9	Incomplete Combustion	--

*Low flue temperature result of incomplete combustion.

TABLE 7

Calculated Mass Flow in Stack from Measurements of Fuel Flow and Excess Air

(Diesel Oil)

Valve Setting	Fuel Flow lb/hr)	Excess Air (%)	Total Air Flow (lb/hr)
5	1.5	125 $\pm$ 12	50 $\pm$ 5
7	3.2	60 $\pm$ 10	65 $\pm$ 5
8	3.8	5 $\pm$ 5	60 $\pm$ 5
9	4.2	(fuel rich)	60

Note: Assume $(A/F)_s = 14.4$

- Mass flow in stack is relatively constant because buoyancy is driving force for air flow.
- Constant air flow causes unacceptable excess air level at low firing rate.
- Air shutter required to maintain constant excess air and minimize efficiency degradation.

C_p = specific heat capacity of flue gas (Btu/lb°F)

T_{F3} = temperature of flue gas (°F)

T_o = temperature of combustion air (°F)

A/F = Air/Fuel ratio

Note that the 72% efficiency at valve setting #9 for fuel oil is the efficiency of the stove and flue pipe as a system.

Gasoline

On gasoline, the turndown was about 5/1 with an input rate of 94 MBtu/hr at valve position #7 (the highest recommended setting for gasoline), and 100-110 MBtu/hr at valve position 8 and 9, which indicates an overfire situation. As the valve setting was increased from 1 to 7, the exit flue temperature increased from 570°F to 970°F and excess air reduced from 230% to 28%. Above valve position #7, the air flow was inadequate for complete combustion, resulting in excessive smoke production and a drop in flue temperature. The lowest measured smoke measurement for gasoline was #5 at valve position 3.

Observations Regarding Startup, Flame Outage, and Safety

Startup. During startup, the stove lid is removed and the fuel is ignited with diesel fuel. The ignition process results in smoke and flue spillage outside the stove through the hole normally covered by the stove lid. This does not cause a safety problem, but can cause obnoxious odors inside the tent.

Igniting the stove when filled with gasoline can be more dangerous to the operator because of the rapid flashing and ignition of gas air mixtures in the stove. However, because the stove lid is removed, an explosion hazard does not exist. The major problem during startup with fuel oil is caused by the poor initial evaporation. When first started, the fuel oil does not evaporate at its required rate because the stove is cold and the pool size of fuel increases to fill the pot to a depth of 1/4". As the stove heats up, it overfires because the large reservoir of fuel begins to be vaporized. Even if valve setting is low, the stove fires as if the setting were at #9, because the entire bottom of the pot is covered with fuel as it would be at the high valve positions. The smoke emissions are very high during this startup transient. The stove operates for about 10-15 minutes in this manner until reaching a steady operating condition which can be controlled by the float valves.

Auto Ignition

If a flame outage occurs, the pool of fuel automatically reignites at valve positions 3-9 because the stove wall temperatures are high enough to vaporize

the fuel and ignite the fuel oil mixture. The reignition is violent and blows the lid off the stove.

Safety. In general, the M41 does not have major safety hazards. The primary hazard stems from simply handling and metering gasoline - a low flash point fuel. To burn gasoline safely, the reliability of the fuel metering could be improved by a double float valve safety.

The pot burner holes tend to clog when using diesel fuel and require frequent cleaning to ensure that they remain open.

Fuel Evaporation Mechanism

To better understand the relationships between the flame size, position, pool size, and fuel evaporation rate, a series of tests were run in which the approximate flame thickness, flame height, and area of the pool were measured. The flame observations were determined by observation through a pyrex view glass installed on top of the stove. The pool area was measured by simultaneously stopping the fuel flow and extinguishing the flame, removing the top of the stove and inner baffle, and measuring the size of the pool. The flame was extinguished by initiating a high flow rate of carbon dioxide into the air inlet. The flow of carbon dioxide into the stove must be maintained even after the flame is extinguished, to prevent auto ignition. Measuring the pool size can be dangerous because of autoignition. An experimenter repeating this test should be protected against possible ignition of the fuel.

The results of this test series are presented in Table 8.

TABLE 8

Fuel Evaporation in Relation to Flame Position and Thickness

Burning Rate		Approximate Flame Thickness	Approximate Flame Height	Area Pool	Thickness (Area Pool) / Height^2	
1.0 lb/hr		2"	4"	9 in^2	1.1	
	4.1x					4.2x
4.1 lb/hr		8"	11 $\pm$ 1	70 in^2	4.6	

- Flame moves away from pool at high firing rates.
- Pool size increases with firing rate.

At an input rate of 20 MBtu/hr, the flame thickness was about 2" and the center of the flame was 4" above the bottom of the pot. The pool area was 9 inches.^{2} When the input rate was increased to 80 MBtu/hr, the flame was about 8" thick and the center of the flame was about 11" above the pot. The pool completely filled the bottom of the pot. These measurements suggest a physical picture for the fuel evaporation.

At steady state, Burning Rate = Evaporation Rate = $\frac{(\dot{q}_{received})}{\Delta H_{vap}}$ Area Pool by necessity.

Assuming radiation is dominant, $\dot{q}_{received} \alpha \frac{(\text{Thickness Flame})}{(\text{Height of flame above pool})^2}$ assuming flame temperature constant.

$$\text{Therefore, Burning Rate } \alpha \frac{(\text{Thickness Flame})\ (\text{Area Pool})}{(\text{Height of flame above pool})^2} \qquad (1)$$

The measurements confirm this physical picture, as shown in Table 4. When input rate was increased by a factor of 4, the (Thickness) (Area Pool)/Height^2 also changed by a factor of 4. The understanding of this mechanism will be very useful in redesigning the stove combustion process.

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